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A Performance Comparison of Secondary Refrigerants

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A PERFORMANCE COMPARISON OF SECONDARY REFRIGERANTS

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ABSTRACT

One potential alternative refrigeration system for supermarket applications is the secondary-loop system. This system uses a conventional direct-expansion refrigeration system to cool a thermofluid that is pumped throughout the supermarket to provide the necessary cooling. In order for this system to be a viable alternative, the loss in performance resulting from the added level of heat exchange and the addition of a pump must be minimized through proper secondary refrigerant selection and system design. Two generalized computer models of a conventional refrigeration system and a secondary-loop system have been developed to aid in this design process. These models were used to investigate the performance of secondary-loop and conventional refrigeration systems. Possible secondary refrigerants are identified. A detailed parametric study of the refrigeration coil and interconnecting pipe diameters was conducted to achieve the best system performance for each fluid and for HCFC-22 in the conventional system. The comparison of these secondary fluids relative to HCFC-22 in a conventional system is presented. The results indicate that through proper secondary refrigerant selection and system design, the secondary-loop refrigeration system could provide equivalent performance to existing systems.

1 INTRODUCTION

Increasing concern in recent years over the environmental effects of chlorofluorocarbon (CFC) and hydrochlorofluorocarbon (HCFC) refrigerants has led to international agreements concerning their phase out. This phase out calls for a stop to production of all CFC's by the year 1996, and an end to production of all HCFC's by the year 2030. The phase out of these refrigerants is of particular concern to supermarkets. Supermarkets are estimated to account for 4% of the national electric power and 30 to 50% of that power is used for refrigeration purposes [McDowell, Klein & Mitchell 1995]. Current supermarket refrigeration systems are direct expansion systems in which each display case contains its own evaporating unit. The direct expansion refrigerant is circulated throughout the supermarket to each of the display cases providing the necessary cooling. A typical system will use on the order of 900 to 1130 kg (2000 to 2500 lb_m) of refrigerant charge. It is estimated that up to 30% of this charge can be lost each year [Harrison, Keeney & Nelson 1995]. This can represent a significant cost to the supermarket. This expense could increase due to the added cost of the hydrofluorocarbon (HFC) mixtures that are expected to replace current refrigerants in most applications. In addition to the cost concerns, there is the added environmental concerns related to the impact of HFC's on global warming.

One potential alternative to using HFC mixtures, is the secondary-loop refrigeration system. This system uses a direct-expansion refrigeration system (primary-loop) to cool a thermofluid. The thermofluid is then pumped throughout the supermarket to provide the necessary cooling. Leakage rates of the primary refrigerant are reduced by containing the direct-expansion system in a compact sealed unit, thereby eliminating the long lengths of interconnecting piping through which the primary refrigerant must flow. It is anticipated that such a system may have a lower overall coefficient of performance (COP) due to the added level of heat exchange and the additional power costs associated with the pump. In order for secondary-loop systems to provide a viable alternative, the loss in performance must be minimized through proper secondary refrigerant selection and system design.

Two generalized computer models have been developed to simulate a conventional system and a secondary-loop system using either volatile or single phase secondary refrigerants. These models were used to compare the performance of heat transfer fluids in the secondary-loop system to a conventional system using HCFC-22 at various capacities. In order to obtain a fair comparison, the size of the refrigeration coil was kept constant. A detailed parametric study of the refrigeration coil geometry, evaporation temperature, and the interconnecting pipe diameter was conducted to determine the best system performance for each fluid at a given capacity. The results of this study are presented here.

2 REFRIGERANT SELECTION

Proper selection of the primary and secondary refrigerants is important in order to achieve the best performance. Because the primary cycle can be contained in a compact sealed unit, more options are available when choosing the primary refrigerant. This includes, in addition to the HFC's, the use of ammonia or hydrocarbons. Ammonia was considered in this investigation as the choice for the primary refrigerant. Stoecker [1989] cites several advantages to using ammonia over HCFC-22, including, a higher cycle efficiency over most temperature ranges and higher heat transfer coefficients. These advantages can be attributed to ammonia's superior thermodynamic and transport properties. Comparing ammonia and HCFC-22 at a given capacity in a conventional direct expansion system, with the same geometry and operating under identical conditions, a 7–12% increase in performance is achieved with ammonia depending upon the capacity chosen.

Unlike the primary refrigerant, it is necessary for the secondary heat transfer fluid to be non-toxic and non-flammable because it is circulated throughout the store. In order to minimize pumping costs, the fluid should also have excellent thermodynamic and transport properties. Because of these restrictions, the available options are limited. There are a variety of commercially developed, single-phase heat transfer fluids which could be used in secondary-loop systems.

- *Propylene Glycol:* Propylene glycol/water solutions are a common heat transfer fluid for medium temperature applications, $T > -20°C(-4°F)$. The freezing point of this solution can be lowered, at the expense of performance, by increasing the percentage of propylene glycol in the solution. A 50% by volume solution can provide freeze protection to about -25°C. The Food and Drug Administration recognizes propylene glycol as a safe food additive. It is non-flammable and non-corrosive to most materials.

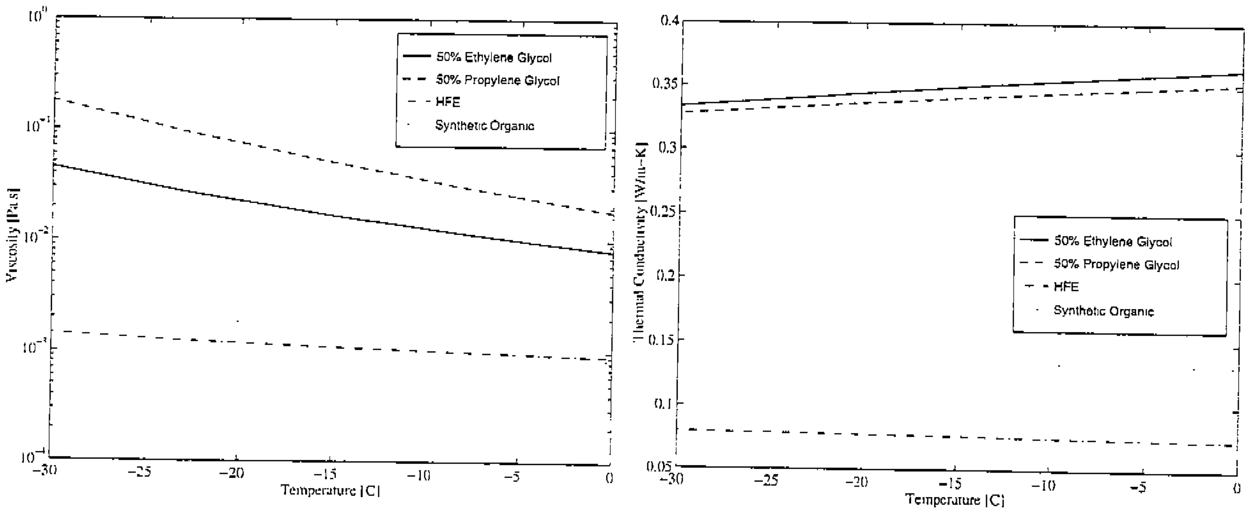
- *Ethylene Glycol:* Ethylene glycol/water solutions provide superior transport properties to propylene glycol solutions and provide a slightly lower freezing point. However, it is orally toxic. As a result, special precautions would be required for use in a supermarket systems. Freeze protection can be obtained to -50°C (-58°F). However, like propylene glycol, lowering the freezing point comes at the expense of decreased heat transfer performance and increased pumping power.

- *Hydrofluoroether:* A new fluid, referred to as a hydrofluoroether (HFE), has recently been introduced. The freezing point of this fluid is listed to be below -100°C, making it suited for both low and medium temperature applications. Preliminary tests have shown the fluid to be orally non-toxic, non-flammable, and it has been found to be compatible with most common materials.

- *Synthetic Organic Fluid:* Several synthetic organic heat transfer fluids are also available. One of these fluids was found to offer freeze protection to -73.3°C (-100°F), thereby providing another option for low temperature systems. This synthetic fluid provides superior low temperature viscosity when compared to ethylene glycol. Like ethylene glycol it is orally toxic and would therefore require special precautions.

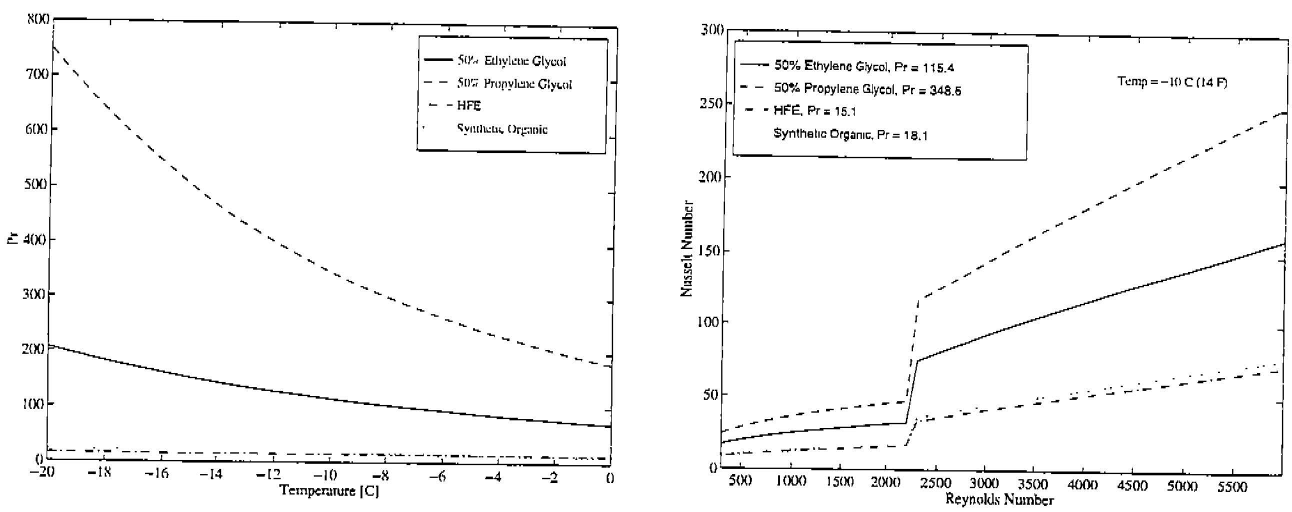
The viscosity and thermal conductivity of each of these fluids versus the temperature is shown in Figure 1. A low viscosity fluid reduces the pumping requirements in the secondary-loop. For a given pipe diameter and fluid velocity, a fluid with a lower viscosity will also have a higher Reynolds number leading to higher heat transfer coefficients. Higher values of the thermal conductivity also lead to improved heat transfer. As can be seen, the HFE and synthetic fluid have superior viscosity values, while the glycols are seen to have higher thermal conductivities. This suggests pumping power may be the critical factor in the performance of the glycols, whereas, heat transfer may be the critical factor for the HFE and synthetic fluid. Figure 2(a) shows the Prandtl number of the fluids versus temperature. Fluids with higher Prandtl numbers can be expected to have higher heat transfer coefficients. The Nusselt number is shown versus the Reynolds number in Figure 2(b) for a constant temperature. The transition between laminar and turbulent flow is seen to occur at a Reynolds number of 2300. The heat transfer coefficient in the laminar regime was calculated with a correlation by Jakob [ASHRAE 1993]. The turbulent heat transfer coefficient was calculated by the Dittus-Boelter correlation [Incropera & DeWitt 1990]. Because the Prandtl number of the glycols is considerably higher, it is possible to have laminar flow and still achieve sufficient heat transfer in the refrigeration coil.

In addition to the above mentioned single phase heat transfer fluids, carbon dioxide as a volatile secondary refrigerant was investigated at low and medium temperature applications. Carbon dioxide has recently received increased attention as a secondary fluid [Hesse 1995, Enkemann & Arnemann 1994, Kauffeld 1995, Pearson & Fellow



(a) (b)

Figure 1: Transport property comparison of single phase heat transfer fluids. (a) viscosity (b) thermal conductivity



(a) (b)

Figure 2: (a) Prandtl number comparison (b) Nusselt number versus Reynolds number at constant temperature

1992/93, Inlow & Groll 1996]. It was a popular refrigerant in the early part of this century, primarily because of its safety. Its use diminished however with the advent of the CFC refrigerants in the 1930's. Enkemann & Arnemann [1994] site several advantages to using a volatile secondary refrigerant, including, higher volumetric heat capacity, constant temperature evaporation, and lower viscosity. The advantages to using CO_2 as compared to using HFCs in a conventional system include an extremely low global warming potential, a high enthalpy of evaporation, it is non-flammable and non-toxic, and its low cost and availability. One drawback to the use of CO_2 is its relatively low critical temperature (31.1°C, 88°F) and high critical pressure (73.7 bar, 1083 psi). The high saturation pressure of CO_2 presents a problem during cycle defrost when the temperatures in the evaporator approach room temperature. Solutions to this problem are given by Enkemann & Arnemann [1994] and Inlow & Groll [1996].

3 SYSTEM MODELS

The secondary-loop and conventional system models predict the capacity and outlet air conditions of the refrigeration coil, as well as, the compressor, fan, and pump (secondary-loop only) power based on a user defined system and operating conditions. There are three main components to each of the models. These are, the refrigeration coil analysis, a thermodynamic analysis of the primary-loop/conventional system, and the analysis of the pressure losses and heat transfer in the interconnecting piping (including the pump in the secondary-loop system model). A multistage compression cycle with flash tank is simulated by the computer models described here. This cycle is necessary in order to allow the systems to produce the low temperatures which are required in supermarket display cases. The economizer cycle was chosen in order to reduce the outlet refrigerant temperatures from the second compressor as much as possible. The refrigeration coil analysis is based on the procedure outlined in Chapter 6 of ASHRAE Equipment volume [ASHRAE 1988]. A detailed description of the models can be found in Inlow & Groll [1996]. The major assumptions used in the models are: a steady-state system, outlet states of the condenser and flash tank are saturated, adiabatic expansion devices, negligible pressure drops in the intermediate heat exchanger and condenser, compressor power is calculated using an isentropic efficiency, condensation occurs over the entire refrigeration coil, and the effects of frost buildup are neglected.

4 PERFORMANCE COMPARISON

The computer models described were used to conduct an investigation of the performance of the secondary-loop and conventional systems at two temperature applications, a medium temperature with an air inlet temperature to the coil of 5.56°C (42°F), and a low temperature with an air inlet temperature to the coil of -20.0°C (-4.0°F). A 50% propylene glycol solution, the HFE, and CO_2 were investigated at the medium temperature. At the low temperature, the HFE, synthetic fluid, and CO_2 were used. A conventional system using HCFC-22 was also modeled at both temperatures. Ethylene glycol was not considered because of its toxicity. The synthetic fluid was considered, despite also being toxic, in order to provide a third secondary refrigerant at the low temperature application. The following general system parameters were kept constant in the analysis:

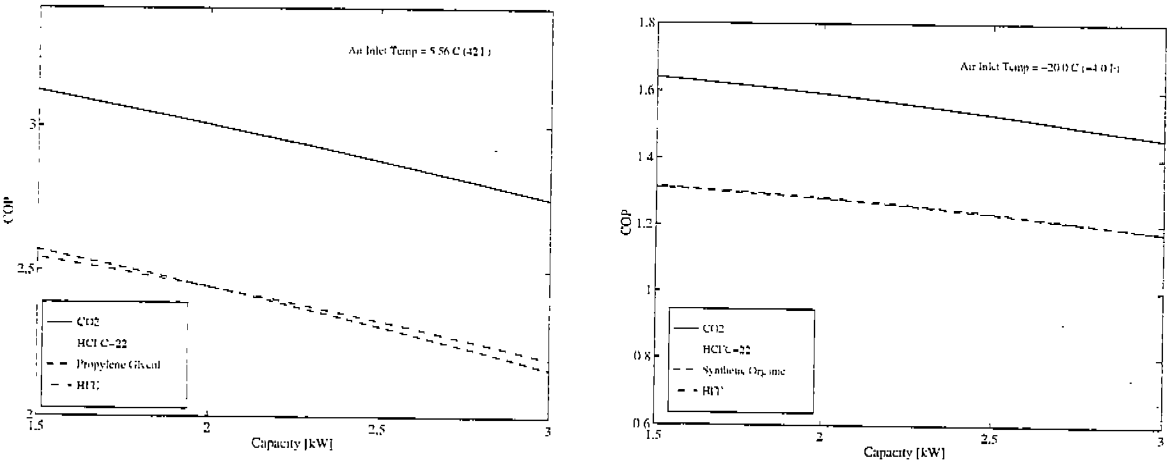
- Coil air face velocity was 1.5 m/s (300 ft/min) with an inlet relative humidity of 50%
- The refrigeration coil used copper tubes in a staggered layout with aluminum fins
- Fin spacing was 78 fins/meter (2 fins/inch)
- 3°C (5.4°F) temperature difference between the outlet refrigerant streams at the intermediate heat exchanger (secondary-loop simulation only)
- Condensing temperature of 40.56°C (105°F)
- Compressor and pump efficiencies were 70%, fan efficiency of 40%
- 60 meters (196 ft) of interconnecting piping with 2.54 cm (1.0 in) of insulation

The fluids were compared at the same capacity and using the same size refrigeration coil. The coil geometry was fixed to be 26.4 cm (10.4 in) long, 15.2 cm (6.0 in) high, and 86 cm (33.9 in) wide.

In order to obtain the best performance with each fluid a detailed parametric study was done on the refrigeration coil and interconnecting piping to determine the optimum value of the coil tube diameter, D_o; the number

Table 1: Secondary-loop geometric parameters

Fluid	D_o [cm]	N_T	N_L	D_{sl} [cm]	D_{ll} [cm]
50% Propylene Glycol	1.59 (5/8 in)	5	10	1.905 (3/4 in)	1.905
Synthetic Fluid	1.59	5	8	1.905	1.905
HFE	1.59	5	8	1.905	1.905
CO_2	1.27 (1/2 in)	6	8	1.27	0.535 (1/4 in)
HCFC-22	1.59	5	8	1.59	0.535



(a) (b)

Figure 3: COP versus refrigeration coil capacity for (a) medium temperature system, (b) low temperature system

of tubes high, N_t, and deep, N_L; inlet refrigeration temperature; suction line (interconnecting piping leading from the coil to the intermediate heat exchanger) diameter, D_{sl}; and the liquid line (interconnecting piping leading from the intermediate heat exchanger to the coil) diameter, D_{ll}. With the coil volume fixed the only way to increase or decrease N_t and/or N_L is to increase or decrease the transverse and/or longitudinal tube spacing. The value chosen for each of these parameters represents a trade-off between increased or decreased heat transfer and increased or decreased pressure drop. Table 1 shows the values of these parameters which were found to provide the highest performance. The optimum inlet refrigerant temperature depends upon the coil capacity. The values of these parameters remain relatively constant for each fluid with the exception of CO_2. Because the vapor density of CO_2 is relatively high, the pressure drop is smaller. Therefore smaller tube diameters can be used.

Figure 3 shows the COP of the secondary-loop and conventional systems versus capacity. The COP is defined as,

$$COP = \frac{Q_{air}}{W_p + W_c + W_f} \tag{1}$$

where Q_{air} is the refrigeration coil capacity, and W_p, W_c, and W_f represent the pump (secondary-loop model only), compressor, and fan power respectively. At the medium temperature, CO_2 and HCFC-22 are within 4% of each other over the entire range. The propylene glycol and HFE provide a similar COP. These two single-phase fluids are within approximately 20% of the HCFC-22 and CO_2. The exceptional performance of the secondary-loop system using CO_2 can be attributed to three factors. One, its high heat transfer coefficients in the refrigeration coil provide a greater capacity at a higher evaporation temperature. The second and third factors are its low viscosity and relatively high vapor density. These two factors reduce pressure losses throughout the system. The lower performance of the single phase fluids can be contributed to the lower heat transfer coefficients and increased pumping power.

At the low temperature, CO_2 provides a definite advantage over the other fluids. Again, the single-phase fluids are similar in performance and are within approximately 20% of the CO_2 system. The performance of HCFC-22 is seen to drop off sharply. This due to the extremely low temperatures which are required to achieve the higher capacities. An evaporation temperature of almost -40°C (-40°F) is required to achieve a capacity of 2.0 kW. The low vapor density of HCFC-22 also leads to high pressure drops in the refrigeration coil, further reducing the COP. The nearly identical performance of the HFE and the synthetic fluid could be expected due to the similarity in the transport properties previously shown. However, unlike the synthetic fluid, preliminary tests have shown the HFE to be non-toxic. This makes the HFE a more favorable heat transfer fluid.

5 CONCLUSIONS

The results presented here show that a secondary-loop system, using CO_2 as a volatile secondary refrigerant and ammonia as the primary refrigerant, provides a COP equivalent to existing systems using HCFC-22. If this system can be designed to overcome the high vapor pressures produced using CO_2, then it could provide a long term, environmental and economic solution for supermarket refrigeration. Secondary-loop systems using single-phase refrigerants do not perform as well due to higher pressure losses and lower heat transfer performance. The COP of these systems could possibly be improved by increasing the refrigeration coil size. This would require cabinet redesign and would reduce product display areas. Despite these shortcomings, secondary-loop systems using single-phase secondary refrigerants would be simpler to design and may be more easily implemented. Further improvements in both secondary-loop systems may still be possible through improvements in the secondary-loop and the primary-loop. Additional secondary refrigerants should also be identified and investigated.

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